ECL- 186A

ENGINEERING CASE LIBRARY

THE PICKUP-HEAD LINK FAILURE (A)

The IBM 2321 is a vast storage device which reads and writes data on a magnetic strip. These strips are stored in cells and mechanically taken from the cells for read-write operations and then returned. With the 2321 in full production and in the field, the pickup-head link began to fail. This link is connected to the head which picks up the magnetic strips and its failure normally damages other components. After a metallurgical fix of the link failed, stresses due to three independent loads were re-examined.

Later the link was redesigned, but the pin on the end of the link began to fail. Further redesign involved metallurgical and structural analysis.

 Prepared by Prof. R. Piziali with support by the San Jose Division of IBM Corporation. Assistance by W. Goddard, J. Minick, D. Olsen, and G. Pastre is gratefully acknowledged.

THE PICKUP-HEAD LINK FAILURE (A)

"Dan, you and Stan have been designated as a task force to correct the 2321 Pickup-Head Link Failure. As you probably know, the "metallurgical fix" was unsuccessful and each time a link goes, we usually lose a drum, a read/write head and a magnetic strip."

The 2321 Data Cell Drive was in full production at this time and many machines had been delivered. Therefore, each failure caused prolonged down time and possible data loss, which is a Cardinal Sin in the data storage business. Therefore, Dan Olsen and Stan Moss would have to work quickly as continued failures of the 2321 would reduce customer satisfaction and could be detrimental to future 2321 sales.

The IBM 2321 Data Cell Drive is a vast capacity, random access storage unit that stores data and retrieves data from magnetic strips (.005" x 2-1/4" x 13") under the direction of a storage control unit. Each strip is one of ten stored vertically in a sub-cell, with 20 sub-cells per cell and ten cells forming a circular array (Exhibit A-1). A strip is taken from the cell, wrapped around a drum (Exhibit A-2a) and passed by a read-write head. The 2321 is designed to be used with IBM computers and in its System/360 and 370 compatible form it can store up to 400,000,000 8-bit bytes per array. The access time to a specific storage location varies from an average minimum of 175ms to an average maximum of 600 ms. These access times are remarkably fast considering the complexity of the mechanical accessing system. When the IBM 2841 Storage Control Unit specifies a storage location, the hydraulic servo of the 2321 rotates the circular array of storage cells until the proper subcell is located at the pickup point. Each strip has two tabs on its upper edge

and a set of separation fingers isolates the desired strip at the pickup point (Exhibit A-2a). A latch keeper (Exhibit A-3c) then releases the torsion spring producing a high acceleration of the drum while a clutch engages the motor drive for continuous rotation (Exhibit A-3b). As the drum rotates clockwise, the pickup head first moves down the chute to pick up the magnetic strip. The continuous rotation then pulls the pickup head and magnetic strip up the chute and on to the drum where the pickup head spring is locked in place (Exhibit A-2b, c, d). This strip is rotated past the read/write head and is then returned to its subcell. The return cycle starts with the compression of a torsional spring (similar to that shown in Exhibit A-3b) which then accelerates the drum counter clockwise as the clutch mechanism reverses the motor drive. The strip and head move down the chute guide, the strip is deposited in its subcell and the pickup head is returned to its ready position (Exhibit A-2d, c, b, a). The return to the ready position compresses the torsional spring (Exhibit A-3b) and sets the latch keeper (Exhibit A-3c) as a trigger.

The pickup head link shown in Exhibit A-2 was the result of three redesigns. The first two designs utilized a locking mechanism on the pickup head to hold the head to the drum during the read/write operation. The following design utilized a locking rod which rotated into place as the pickup head reached the drum (Exhibit A-2). The hook on this arm was later replaced with a spring and the final link and spring designs are shown in Exhibit A-4.

The manufacture of the pickup head link started with an AISI 8620 steel forging. After both its rough machining and finish machining, the link had to be straightened as it become slightly twisted due to the

releasing of internal stresses. Once the proper dimensions were attained, all but the tip area was plated with copper. The link was put into a carburizing oven, heated and quenched, carburizing the uncoated tip to produce a wear surface. The copper is then chemically removed from the link and replaced with a chrome plating for corrosion resistance.

When failures were first observed (see Exhibit A-4, assembly drawing), the links were taken to IBM's metallurgy group for analysis. Metallurgy reported that some surface decarburization was observed in the area of fracture. A decarburization indicates that carbon has gone out of solution, thus softening the metal. The link design engineer decided that a carbon enrichment process would bring the link surface back to its original carbon content and strength and eliminate the failures. With this change, new links went into production and testing. However, just after these new links were released they began to fail.

With the preceding background, Dan and Stan decided to start their investigation by recalculating the expected stresses at the link break point. They considered three loading sources; the spring load, the inertia load and the shock load. The spring load is generated when the link spring is locked to the drum. The calculation of this force existed in a previous 2321 report and Dan and Stan were satisfied with the 3.8 lb. maximum force listed. A report also existed comparing theoretical inertial forces with experimental results. The theoretical results predicted a maximum acceleration of 54.2 g's on the 12.4-gram head and 3.0-gram magnetic strip. Experiments were run with a 30-inch mylar strip instrumented first with strain gauges and then with an accelerometer. The signals identified one acceleration peak as the head picked up the mylar

strip and started back up the guide and another as the head hit the drum. Only the first peak is reflected as a load on the links and it was found to be 52.8 g's and 29 g's in the two experiments. Dan and Stan decided to use 54.2 g's as their design acceleration.

The final load source was shock loading at the cycle start and at latch. At the cycle start the high clockwise acceleration of the drum slams the pickup head against the outside of the guides. The friction between the pickup head shoes and the guides produces an impact force on the link. At the end of the cycle the pickup head deposits a strip into its subcell, and returns to the ready position (Exhibit A-2b to A2a). During its upward motion, the link is at a severe angle with the guides and friction between the pickup head shoes and the guides will load the links. However, a more severe load occurs at latch. As the drum reaches the end of its travel, the index roller rotates the lever compressing the torsion spring (Exhibit A-3b). At the latch (Exhibit A-3c) the latch striker moves past the spring-loaded latch keeper and impacts the latch bumper (the bumper is a safety over-ride feature). The drive shaft then reverses direction and is restrained by the latch keeper. However, the overshoot during bumper compression causes the pickup head to move further up and out against the guide. As the latch striker rebounds from the bumper, the pickup head shoes are against the outside of the guide and absorb the rebound forces. Occasionally, the link-guide angle exceeds the friction angle and the pickup head shoes lock in the guides, resulting in a system malfunction. It is almost impossible to accurately calculate these shock loads and experimental values have to be used in stress calculations.

Dan and Stan checked with the link designer and found that he was using the same spring and inertia forces that they had accepted and he was using a shock load of 15 lbs. The designer had taken this shock load from a single experimental result whose source had become somewhat obscure by this time. Stan had been involved in the original design of the 2321 links and his rather rough experiments had measured shock loads as high as 30 lbs. Many changes had occurred in the 2321 since Stan's experiments but he felt that 15 lbs was too low and that new experiments should be run to determine the shock loading in the present system. Dan assumed the responsibility for these experiments while Stan calculated the stress in the links using his old data of 30 lbs shock loading per link. Before Stan could start his stress calculations he had to determine the eccentricity at which the load was applied to the link. The eccentricity existed because all of the loading was transmitted through the link pin. By observing wear marks on the pins of broken links, Stan decided to use a loading eccentricity of .125 inch between the load point and the center of the links .062-inch depth.

Questions:

1. Calculate the stress at the normal breaking point (Exhibit A-4) with the loading and load point assumed by Stan. Is the link strong enough to withstand the loads? If not, design a link that can withstand these loads.

2. Calculate the stresses in the spring and determine its safety factor.

3. Are there any other areas which appear to be critical stress points? Are they conservatively designed?

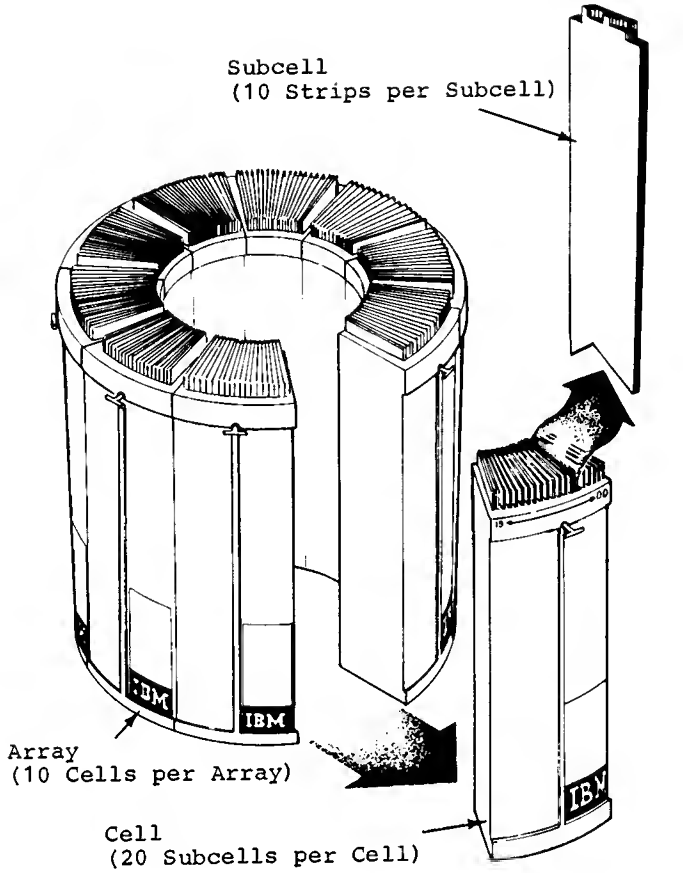


EXHIBIT A-1

IBM 2321 Drive, cell and subcell.

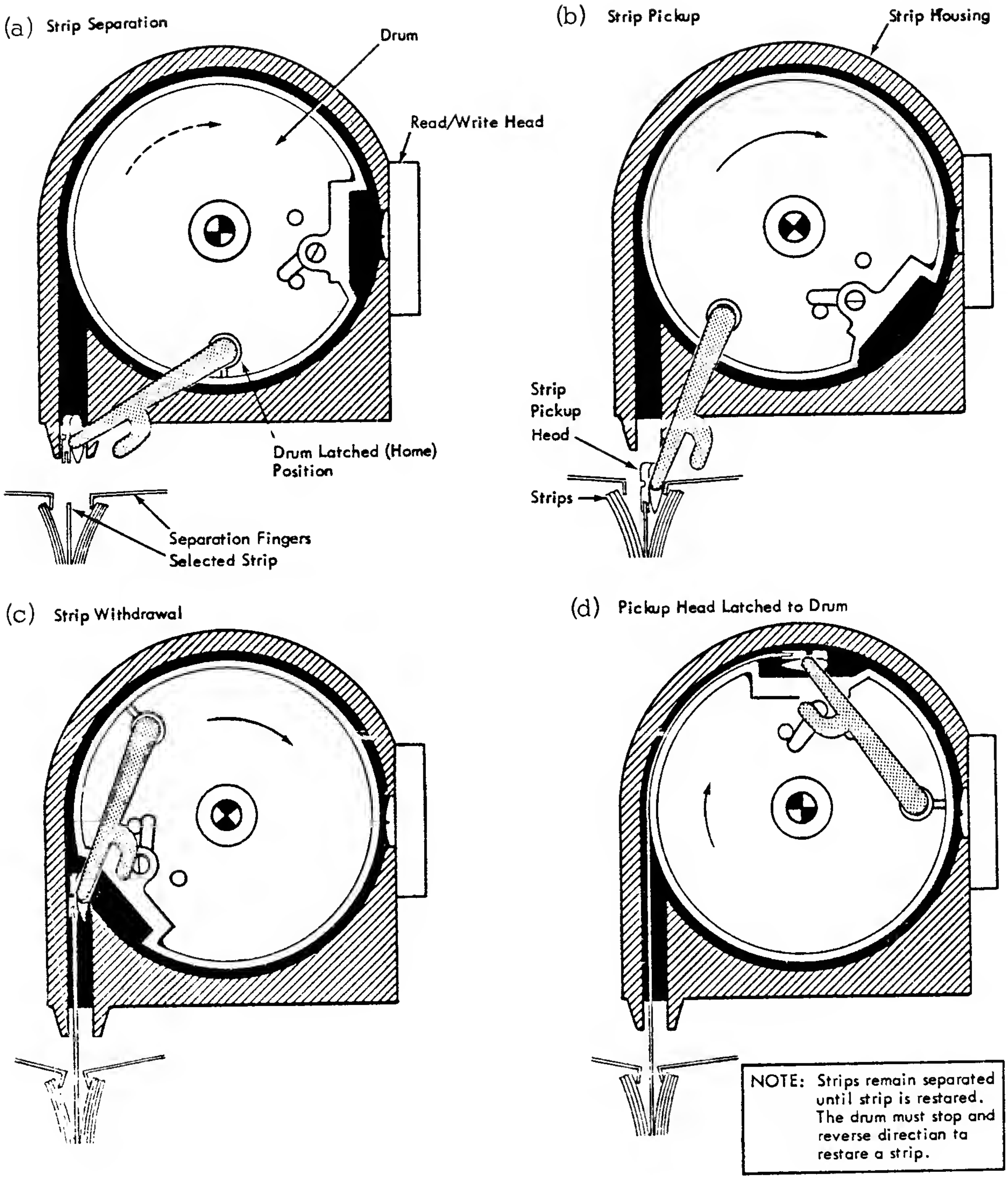
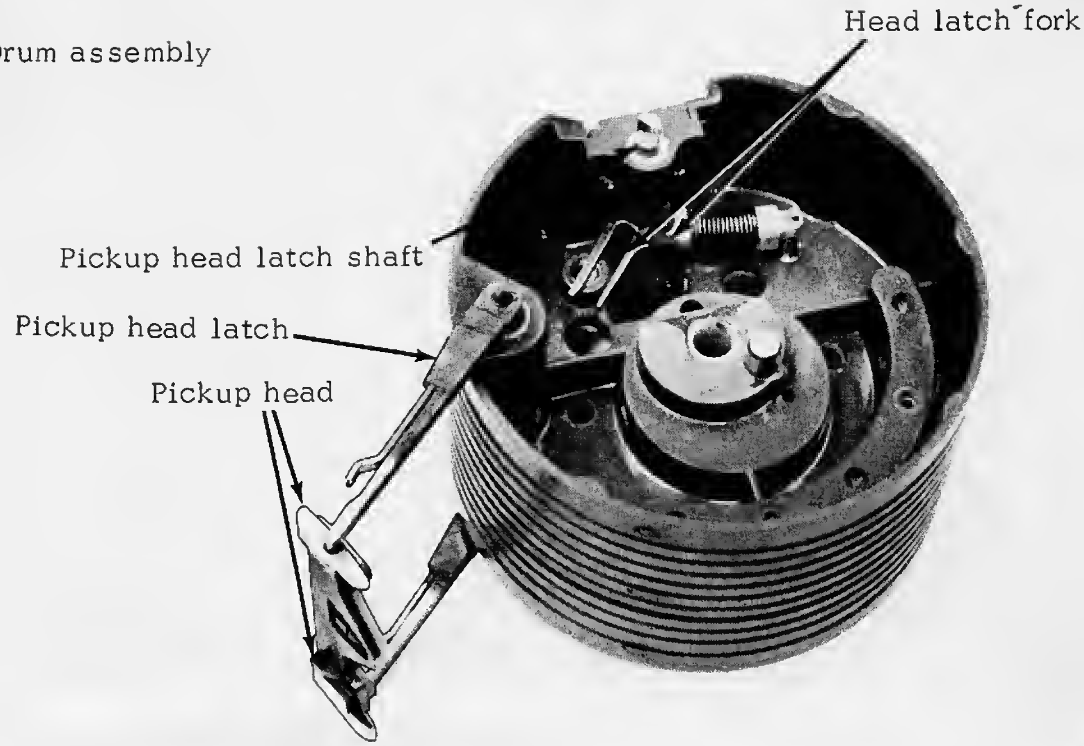


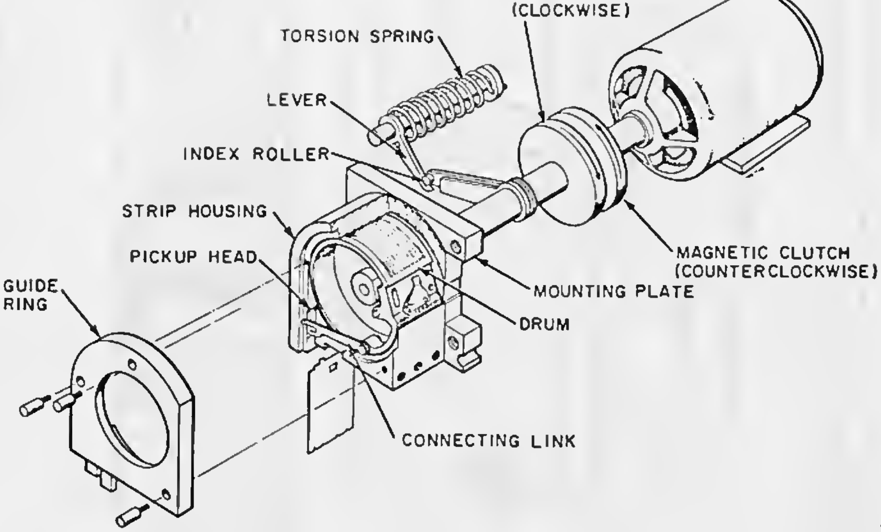
EXHIBIT A-2

Pickup mechanism cycle.

(a) Drum assembly



(b) Drum drive



The latch striker is mounted on the main drive shaft, although this is not shown in (b).

(c) Latch keeper

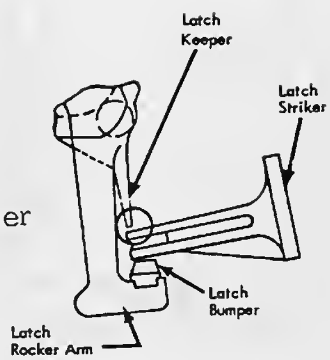


EXHIBIT A-3

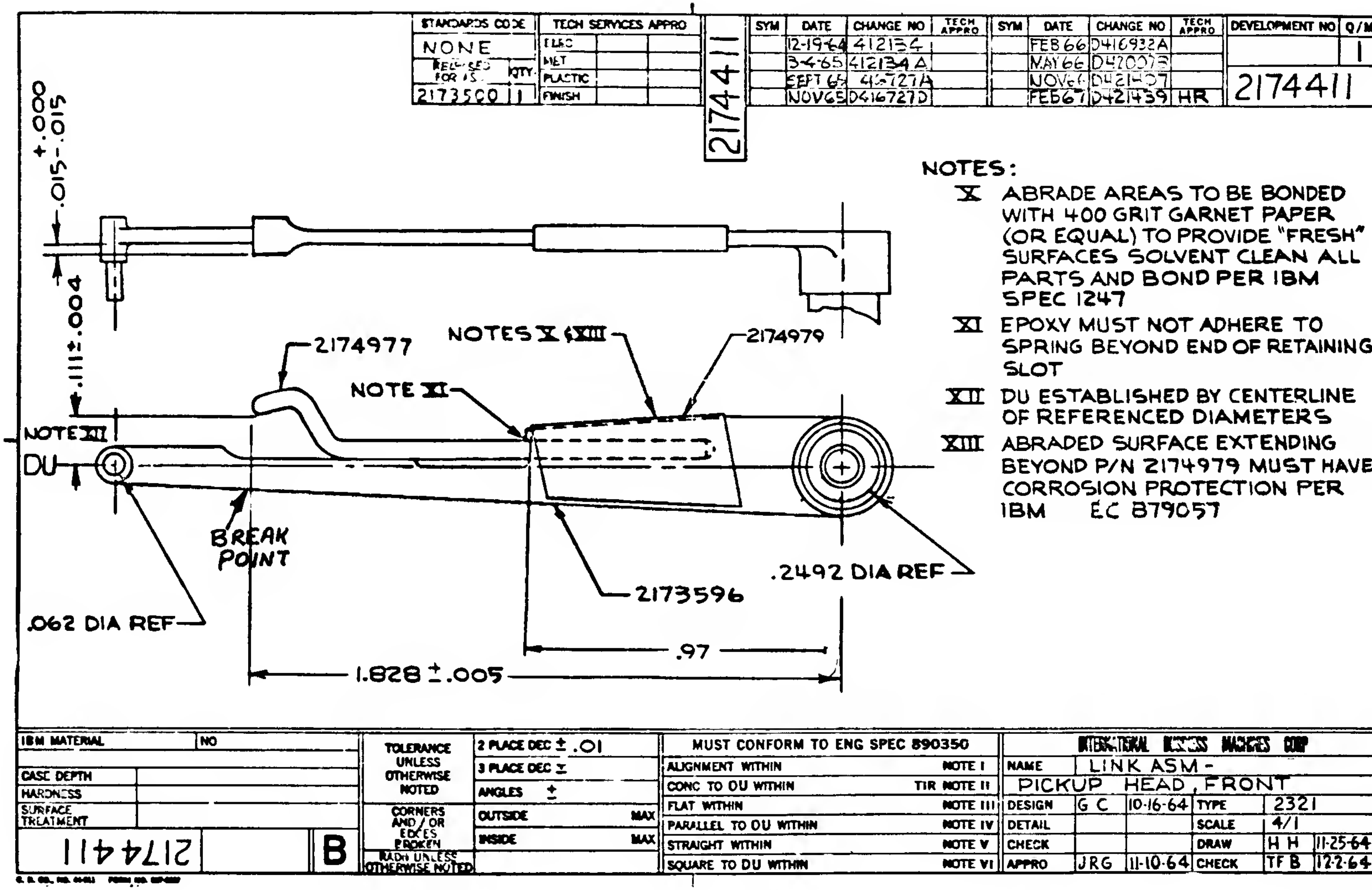

STANDARDS CODE
NONE
RELEASED FOR IS
QTY
2173500 11
TECH SERVICES APPRO
ELEC
MET
PLASTIC
FINISH
2174411
SYM
DATE
CHANGE NO
TECH APPRO
12-19-64 412134
3-4-65 412134 A
NOV65 D416727D
FEB 66 D416932A
MAY 66 D42007
FEB67 D421439 HR
DEVELOPMENT NO
Q/M
2174411
NOTES:
X ABRADE AREAS TO BE BONDED WITH 400 GRIT GARNET PAPER (OR EQUAL) TO PROVIDE "FRESH" SURFACES SOLVENT CLEAN ALL PARTS AND BOND PER IBM SPEC 1247
XI EPOXY MUST NOT ADHERE TO SPRING BEYOND END OF RETAINING SLOT
XII DU ESTABLISHED BY CENTERLINE OF REFERENCED DIAMETERS
XIII ABRADED SURFACE EXTENDING BEYOND P/N 2174979 MUST HAVE CORROSION PROTECTION PER IBM EC 879057
+.000
.015-.015
.111±.004
2174977
NOTES X & XIII
2174979
NOTE XI
NOTE XII
DU
BREAK POINT
.062 DIA REF
2173596
.2492 DIA REF
.97
1.828 ±.005
IBM MATERIAL
NO
CASE DEPTH
HARDNESS
SURFACE TREATMENT
2174411
B
TOLERANCE UNLESS OTHERWISE NOTED
2 PLACE DEC ± .01
3 PLACE DEC ±
ANGLES ±
CORNERS AND/OR EDGES BROKEN
OUTSIDE MAX
INSIDE MAX
RADII UNLESS OTHERWISE NOTED
MUST CONFORM TO ENG SPEC 890350
ALIGNMENT WITHIN NOTE I
CONC TO DU WITHIN TIR NOTE II
FLAT WITHIN NOTE III
PARALLEL TO DU WITHIN NOTE IV
STRAIGHT WITHIN NOTE V
SQUARE TO DU WITHIN NOTE VI
INTERNATIONAL BUSINESS MACHINES CORP
NAME LINK ASM - PICKUP HEAD, FRONT
DESIGN G C 10-16-64
TYPE 2321
DETAIL
SCALE 4/1
CHECK
DRAW H H 11-25-64
APPRO JRG 11-10-64
CHECK TF B 12-2-64

EXHIBIT A-4

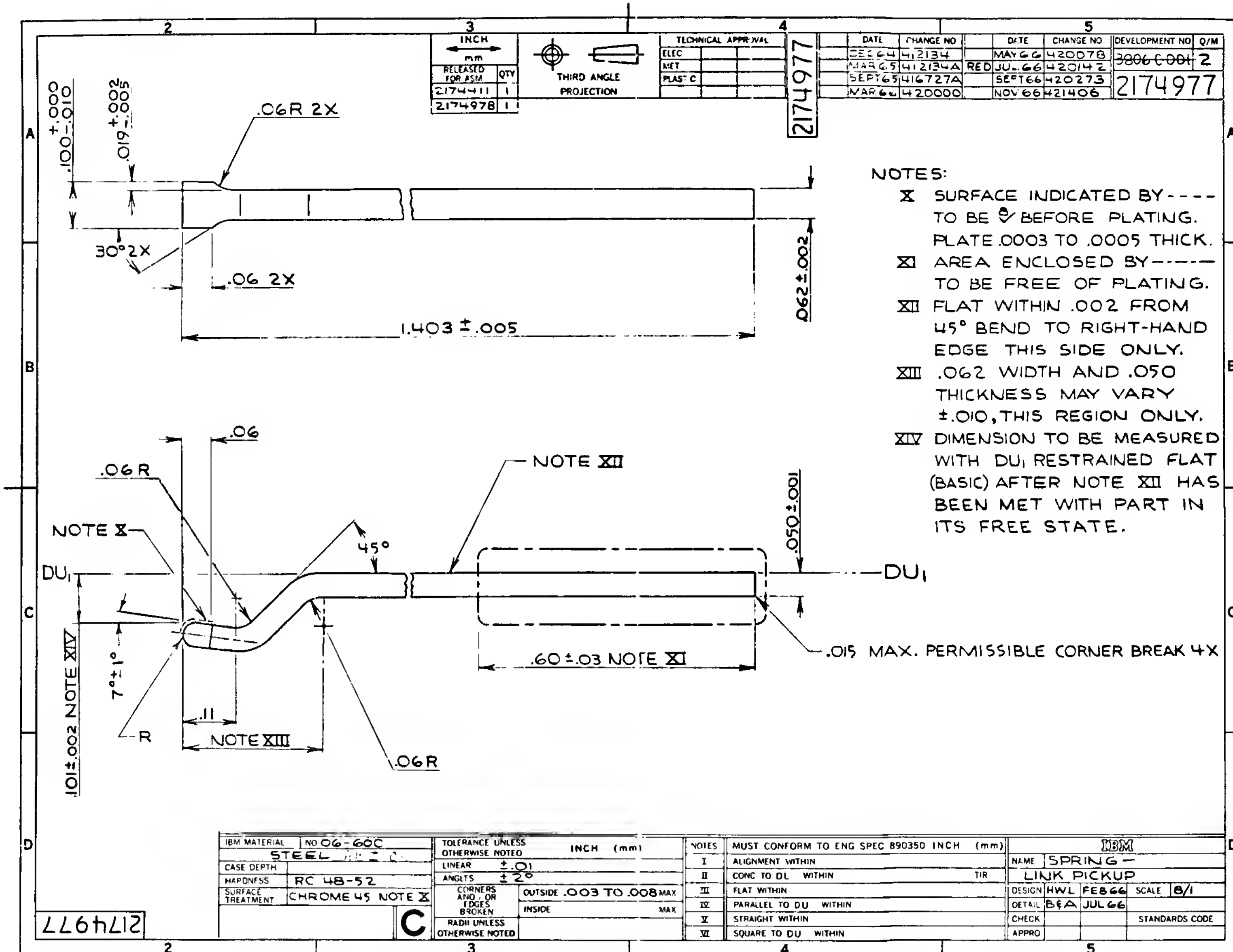

NOTES:
X SURFACE INDICATED BY - - - - TO BE ∇ BEFORE PLATING. PLATE .0003 TO .0005 THICK.
XI AREA ENCLOSED BY – - – - – TO BE FREE OF PLATING.
XII FLAT WITHIN .002 FROM 45° BEND TO RIGHT-HAND EDGE THIS SIDE ONLY.
XIII .062 WIDTH AND .050 THICKNESS MAY VARY ±.010, THIS REGION ONLY.
XIV DIMENSION TO BE MEASURED WITH DU1 RESTRAINED FLAT (BASIC) AFTER NOTE XII HAS BEEN MET WITH PART IN ITS FREE STATE.
.06R 2X
.100+.000 -.010
.019+.002 -.005
30° 2X
.06 2X
1.403 ± .005
.062 ± .002
.06
.06R
NOTE X
NOTE XII
45°
DU1
.050 ± .001
.015 MAX. PERMISSIBLE CORNER BREAK 4X
.60 ± .03 NOTE XI
7° ± 1°
.101 ± .002 NOTE XIV
R
.11
NOTE XIII
.06R
THIRD ANGLE PROJECTION
RELEASED FOR ASM 2174411 2174978
DEVELOPMENT NO 2174977
IBM MATERIAL NO 06-60C STEEL
HARDNESS RC 48-52
SURFACE TREATMENT CHROME 45 NOTE X
TOLERANCE UNLESS OTHERWISE NOTED LINEAR ±.01 ANGLES ±2°
CORNERS AND/OR EDGES BROKEN OUTSIDE .003 TO .008 MAX
MUST CONFORM TO ENG SPEC 890350
IBM
NAME SPRING – LINK PICKUP
DESIGN HWL FEB66 SCALE 8/1
DETAIL B&A JUL66
2174977

EXHIBIT A-4

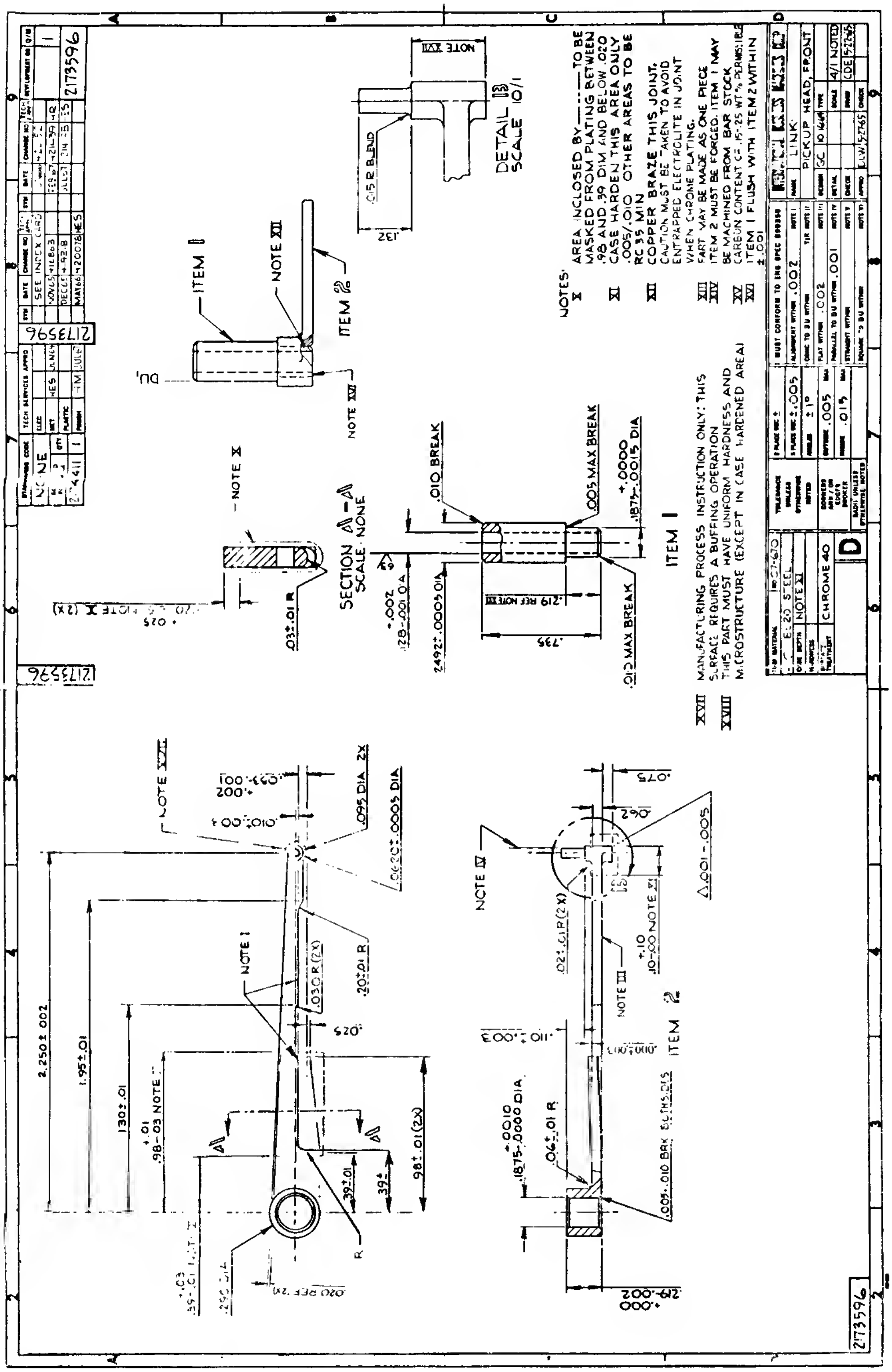

2173596
DETAIL B
SCALE 10/1
NOTE XVII
.015 R BLEND
.132
ITEM 1
NOTE XII
ITEM 2
NOTE XV
SECTION A-A
SCALE: NONE
NOTE X
.010 BREAK
.005 MAX BREAK
.010 MAX BREAK
.735
.219 REF NOTE XIII
ITEM 1
NOTES:
X AREA INCLOSED BY ---- TO BE MASKED FROM PLATING BETWEEN .98 AND .39 DIM AND BELOW .020
XI CASE HARDEN THIS AREA ONLY .005/.010 OTHER AREAS TO BE RC 35 MIN
XII COPPER BRAZE THIS JOINT. CAUTION MUST BE TAKEN TO AVOID ENTRAPPED ELECTROLITE IN JOINT WHEN CHROME PLATING.
XIII PART MAY BE MADE AS ONE PIECE
XIV ITEM 2 MUST BE FORGED. ITEM 1 MAY BE MACHINED FROM BAR STOCK
XV CARBON CONTENT OF .15-.25 WT % PERMISSIBLE
XVI ITEM 1 FLUSH WITH ITEM 2 WITHIN ±.001
XVII MANUFACTURING PROCESS INSTRUCTION ONLY: THIS SURFACE REQUIRES A BUFFING OPERATION
XVIII THIS PART MUST HAVE UNIFORM HARDNESS AND MICROSTRUCTURE (EXCEPT IN CASE HARDENED AREA)
E 20 STEEL
CHROME 40
LINK
PICKUP HEAD, FRONT
D
2.250 ± .002
1.95 ± .01
1.30 ± .01
.98 ± .01 (2X)
.39 ± .01
.095 DIA 2X
NOTE I
NOTE III
NOTE IV
ITEM 2
.005-.010 BRK EDGES
.001-.005

EXHIBIT A-4

ENGINEERING CASE LIBRARY

ECL 186B

THE PICKUP-HEAD LINK FAILURE (B)

THE PICKUP-HEAD LINK FAILURE (B)

When Dan Olsen and Stan Moss discovered that the link design engineer had used 15 lbs as the maximum shock load, they decided to check this assumption as their first attempt to solve the link failure. Their strategy was for Stan to analyze the link stress assuming his old experimental value of 30 lbs shock load while Dan set up an experiment to determine the actual shock loads in the present system. The calculations would be completed well before the experiment and, if they indicated that overstressing would occur at 30 lbs, Dan would continue the experiments. However, if overstressing did not appear to be the cause of failure, Dan and Stan would have to look for other possibilities. The three link loads were the spring load, inertial load, and shock load. These occurred at separate times in the cycle, and they could be handled independently. Stan used a beam-bending model for the spring load and an eccentrically loaded column model for the inertia and shock loads. He calculated stresses of 29,800 psi, 28,600 psi and 144,000 psi, respectively. Data from the materials analysis group indicated that one could expect an ultimate strength of 155,000 psi and an endurance strength of about 70,000 psi (Exhibit B-1). Shock loads of 30 lbs could cause failure either by direct overloading or by fatigue. These calculations were completed just a couple of days after the task force was formed and are summarized in Stan's memo (Exhibit B-1).

The urgency of eliminating the link failures required an immediate redesign on the basis of Stan's calculations. Meanwhile, work on the experiments would continue to determine the validity of the assumed shock load. Stan's memo recommended that the minimum width and depth be increased to .100 inch, thereby reducing the shock induced

stress to 28,000 psi. Dan and Stan wanted to increase the width on the inside of the link as this would be the most efficient use of material. It would reduce the eccentricity of loading and increase the strength of the cross section. However, system tolerances indicated that such an addition could result in interferences and the material had to be added to the outside of the link.

Attempts to add material to the height of the link were unsuccessful as interferences would result for any increase in this dimension. A change of material had been considered and rejected because of the necessity of carburizing the pin area. The new link design is shown in Exhibit B-2. During his calculations Stan had also found that the link spring strength was marginal. Therefore, his design called for a spring of .100 inch in width to match the width of the new link and to provide an adequate safety factor. During the week in which Dan and Stan were conducting their tolerance investigation to determine where they could add material, they had the model shop rough machine a few links. Once the final dimensions were established, these links were finished machined, treated and put into life tests, less than two weeks after the investigation had begun.

Four weeks after the task force formation, Dan completed the shock load tests and found normal shock loads of 22.4 lbs and worst case shock loads of 47.7 lbs. The loads at the cycle start and at latch-up were approximately equal.

Dan and Stan were now convinced that overload had caused the link failures, but they decided to extend their life tests before starting

the manufacture of new links. Two weeks later, with approximately 10 million cycles on the links (twice their design life) the new design was released for manufacture and immediate replacement of the old links.

With the link problem solved, Dan became concerned that he may have created a new problem. It was possible that the increased link stiffness would increase the pin shock loading and cause pin failures (Exhibit B-2, Detail C). Dan set up an experiment with the strain gauges mounted as close to the pin as possible and found that the new link did not create a measurable increase in force. Dan surmized that the flexibility of the system made the increased link stiffness insignificant.

About one year later, Dan began to question these conclusions as the link pins began to break off. The pins were taken to IBM's metallurgy group who analyzed the problem as a fatigue failure. Dan redid his stress calculations and concluded that the pin was overdesigned by a safety factor of 3 or 4 and could not be overstressed. Furthermore, if the pins were overstressed a major redesign would be required. The pins are inserted into the pickup head shoe and the cross section at that point was as thin as possible. An increase in the pin diameter would require an increase in the shoe height and therefore an increase in the guide and drum housing dimensions.

Dan tried various ways of analyzing the problem, but the pin always came out with a reasonable safety factor. Also confusing was the fact that the pins failed in the uniform section, just above the .02 in. blending radius rather than at the blending radius stress concentration (Exhibit B-2, Detail C). Metallurgy did not find any machining marks

which could act as stress concentrations, but continued to identify the failure as one of fatigue. Dan refused to accept the failure as one of overloading and a professor from San Jose State was brought in as a consultant. After examining the broken pins, the professor concluded that overloading could not have been the cause of failure because the areas of initial failure were much too small. Dan smiled. Upon close examination, Metallurgy found white areas about .0001 inches thick near the break point. These flakes indicated surface decarburization which would result in soft spots that could serve as failure initiation points. Therefore the failure had been one of fatigue, but it was due to a metallurgical imperfection rather than to overloading.

Dan immediately had some pins buffed to remove the surface decarburization and put the parts into life test. When the pins did not fail, the buffing operation became part of the link manufacturing process. Dan also tightened tolerances in the pin areas to prevent binding or eccentric loading and he changed surface specifications in the pin area, requiring increased smoothness to eliminate stress risers and shot peening to introduce compression in the surface (see Exhibit B-2, Notes XVII and XX).

During the following years, Dan has come to the conclusion that a change from AISI 8620 steel to a maraging steel would have been the most elegant solution to both of his problems, but unfortunately, maraging steel had not been developed at the time of his redesign. Since both the pin and the link have now been operating successfully in the field for over four years, no further changes have been made.

ECL 186B

SYSTEMS DEVELOPMENT DIVISION
2321 Technical File Memo No. 178
Department 567, San Jose
By: Mr. S. K. Moss
June 29, 1967

SUBJECT: Pickup Head Link Failure

I. Introduction:

The failure of the pickup head links is catastrophic. There is almost certain damage to strip, housing, and magnetic head when one breaks. A number of failures in the field and in testing stimulated this analysis to determine the nature of the failures and suggest a solution.

II. Conclusions:

The links are failing in "fatigue as eccentrically loaded columns from compression and tension shock loads during drive latch up. Conditions are aggravated by shoe wear, pivot bushing wear, and latching hard.

III. Recommendations:

Increase the link section to .100"wide, and reshape to a minimum height of .100". The increase in width is more important to correct the failure.

IV. Discussion:

Three loads were considered in the stressing of the links. They are:

1) The spring - which holds the pickup head to the drum. This force places the link in bending in the direction of the larger section. A load of 3.8 lbs. was used as a maximum for calculations.

2) The inertia - of the pickup head and strip as it accelerates and decelerates up and down the straight portion of the chute.
This force places compression and tension loads on the link, but it is applied eccentrically which also causes bending in the direction of the smaller section of the link. A value of 54.2 g's was used for calculations.

EXHIBIT B-1

3) The shock - loads at latch up. One of the last functions of the drive is to lift the pickup head up the chute away from the strips. At this point the links assume a very severe angle with the pickup head guides and due to the friction between the guides and the pickup head shoes, there is considerable force, tension and bending, in the links. As the sequence of action continues, the drive strikes the latch bumper and rebounds, slamming the pickup head against the opposite side of the guides causing severe compression and bending loads in the link. The "stick-slip" nature of the friction, with the materials and load angle involved, causes a stressing fluxion which appears as repeated shock loads. This is the time of maximum stressing of the links. Some maladjustment and unfavorable tolerance conditions can cause the pickup head to hang up in the guide when the links exceed the maximum angle of the coefficient of friction. There has even been a term assigned, known as "straight arming" the drive. This load is applied to the link eccentrically as in the case of the inertia. The start of the cycle also throws a compression and bending load onto the links. A value of 30 lbs. was used for calculations (see Mars Technical File Memo=103 by S. K. Moss. The tests were repeated in June 1967. The results were that with an eccentricity of .125 there is a force of 22.4 lbs. under normal operation and a worst case of 47.7 lbs.)

This problem was approached as though the links were slender columns. First it was noted by using Euler's formula, that if you had a slender column, with uniform cross section equal to the smallest cross section of the link, (.080 x 0.62), round ended, and with an unsupported length equal to the link pivots (2.25"), the column would fail when stressed to 18,700 lbs./in.^{2}. This is not the case of the link, of course, but it does give us a clue, and the link is uniform in the direction of the least radius of gyration. In addition the link is loaded eccentrically such that it will bend about the axis of the least radius of gyration. Some data was taken loading a link eccentrically in compression noting deflection and behavior. The link appears to act as a column with one end wholly fixed and the other end free to move laterally as well as rotate. The eccentricity of loading was determined to be .125 inch as indicated by wear marks. Note that the eccentricity of loading will increase as the

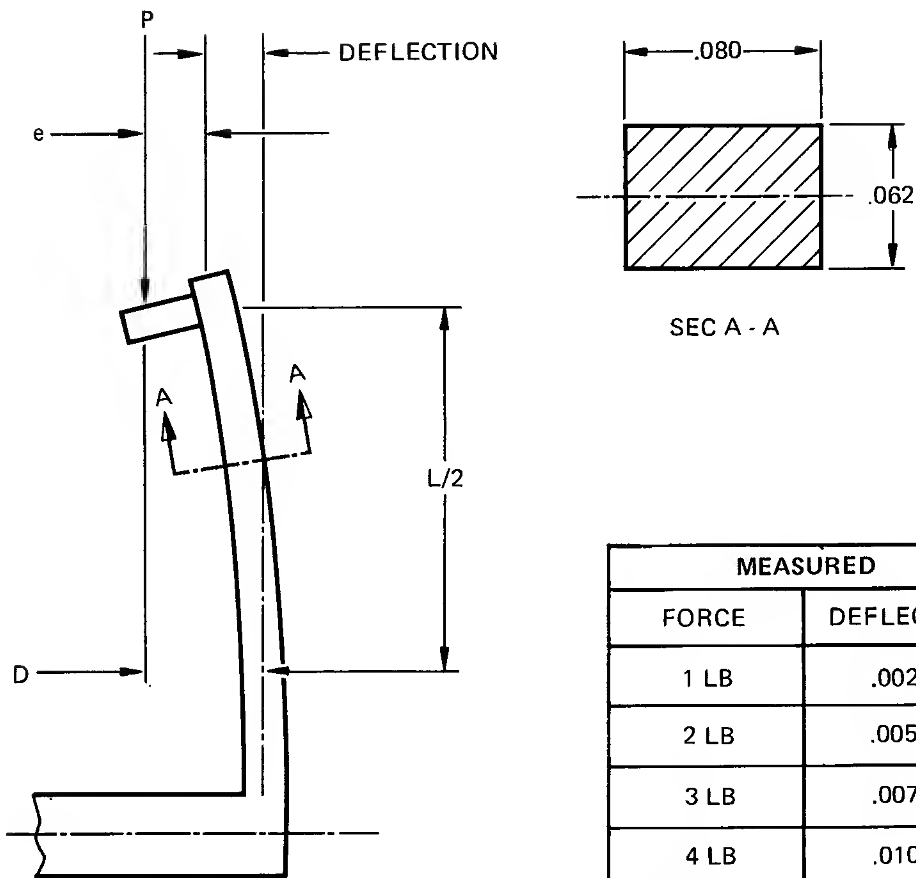
bushing in the pickup head wears, (bellmouths). The unsupported length of the column appears to be just a little more than twice the slender portion of the link (L = 2.625"). This is concurrent with the end conditions described, as the link assumes the shape of a cantilevered beam with a load at the end. Then it becomes apparent that this is 1/2 of an equivalent column with both ends free to rotate. Accepting these conclusions, the deflections calculated match fairly well with the data taken experimentally. Then with these parameters the Secant Formula for columns with eccentric loads was applied. Using the 30 lb. shock load figure, the stress was calculated to be <u>144,000</u> lbs/in^2. Personnel from the Materials Analysis group stated that the link material (8620) under similar conditions of hardness and size will have an endurance stress limit on the order of 70,000 lbs/in^2 and an ultimate tensile strength of 155,000 to 160,000 lbs/in^2. It was then concluded that the links are failing in "fatigue" as eccentically loaded columns. An explanation of why the column does not continue to collapse when stressed so highly is that the system is stopped by the striker.

It has been proposed to strengthen the link by adding .038" to the outside and .022" to the top bringing the smallest section to .100" x .100". Everything else being equal, the stress will be reduced to <u>28,000</u> lbs/in.^{2}. Calculations indicate that the inertia load in the current link design causes a stress of 28,600 lbs./in.^{2} and the hold down spring 29,800 lbs./in.^{2}.

Technical File Memo No. 178

APPENDIX

SAMPLE CALCULATIONS



MEASURED	
FORCE	DEFLECTION
1 LB	.0025"
2 LB	.005"
3 LB	.0075"
4 LB	.010"
4.5 LB	.013"

e = .125

EXHIBIT B-1

Technical File Memo No. 178

$$I = \frac{b\,h^3}{12} = \frac{.080 \times (.062)^3}{12} = 15.9 \times 10^{-7}$$

$E = 30 \times 10^6$ if $\frac{L}{2} = 1.312$ (Tip to end of slot = 1.27)

$P = 4\#$

$D = e \times \sec\left(\frac{PL^2}{4EI}\right)^{1/2}$ - Mechanics of materials 2nd
Laurson & Cox
Sec 165. Pg. 257

$$D = .125 \sec\left(\frac{4 \times 2.625^2}{4 \times 30 \times 10^6 \times 15.9 \times 10^{-7}}\right)^{1/2}$$

$$D = .125 \sec\sqrt{.145}$$

$.381 \text{ rad} = .381 \times 57.3 = 21.8°$

$\sec 21.8° = 1.0770$

$D = .125 \times 1.0770 = .1348$

Deflection $= .1348 - .125 =$ <u>.0098</u>

Secant Formula

$S = \frac{P}{A}\left(1 + \left(\frac{ec}{r^2}\right) \sec \frac{L}{r}\sqrt{\frac{P/A}{4E}}\right)$ Mechanics of Materials 2nd Ed.
Laurson & Cox
Sec 165. Pg. 258

EXHIBIT B-1

$$r = \frac{h}{12} = .289 \times .062 = .0179 \text{ in.}$$

P = 30 lbs (shock) e = .125

L = 2.625 in. A = .00496 in^2

C = .031

$$\frac{P}{A} = \frac{30}{.00496} = 6{,}050 \text{ lbs/in}^2$$

$$\frac{ec}{r^2} = \frac{.125 \times .031}{.0179^2} = 12.1$$

$$\frac{L}{r} = \frac{2.625}{.0179} = 146.5$$

$$\sqrt{\frac{P/A}{4E}} = \sqrt{\frac{6{,}050}{.4 \times 30 \times 10^6}} = 7.11 \times 10^{-3}$$

$$\text{rad} = 146.5 \times .00711 = 1.041$$

$$1.041 \times 57.3 = 59.6°$$

$$\sec 59.6° = 1.9762$$

$$S = 6{,}050\,(1 + 12.1 \times 1.9762)$$

$$= \underline{\underline{144{,}200}} \text{ lbs/in}^2$$

EXHIBIT B-1

Technical File Memo No. 178

INERTIA LOAD - 54.2 g's

Head = 12.4 grams Strip = 3.0 grams

$W = 15.4 \times 2.205 \times 10^{-3} \times 54.2 = 18.4$ lbs.

$P = W$ Each link $= \frac{18.4}{2} = 9.2$ lbs.

$\frac{P}{A} = \frac{9.2}{.00476} = 1{,}855 \ \text{lb/in}^2$ $\qquad \frac{L}{r} = 146.5$

$\sqrt{\frac{P/A}{4E}} = \sqrt{\frac{1{,}855}{4 \times 30 \times 10^6}} = .00394$ $\qquad \frac{ec}{r^2} = 12.1$

Rad = 146.5 x .00394 = .578 = 33.1°

Sec 33.1° = 1.1937

$S = 1{,}855\,(1 + 12.1 \times 1.1937)$

$= \underline{\underline{28{,}600}} \ \text{lbs/in}^2$

SPRING LOAD

Max Spring = 3.8#

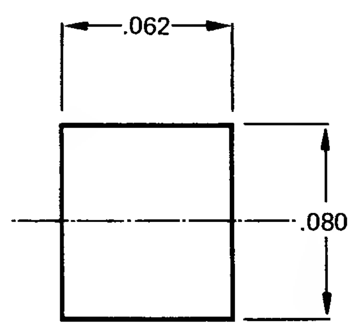


EXHIBIT B-1

Technical File Memo No. 178

$$I = \frac{b\,h^3}{12} = \frac{.062 \times .080^3}{12}$$

$$S = \frac{Mc}{I} = .00000264$$

$$C = .040$$

$$M = 3.8 \times .500 = 1.9 \text{ in. lb.}$$

Distance to breaking area

$$= \frac{1.9 \times .040}{2.64 \times 10^{-6}}$$

$$= \underline{\underline{29,800}} \text{ lb/in}^2$$

EXHIBIT B-1

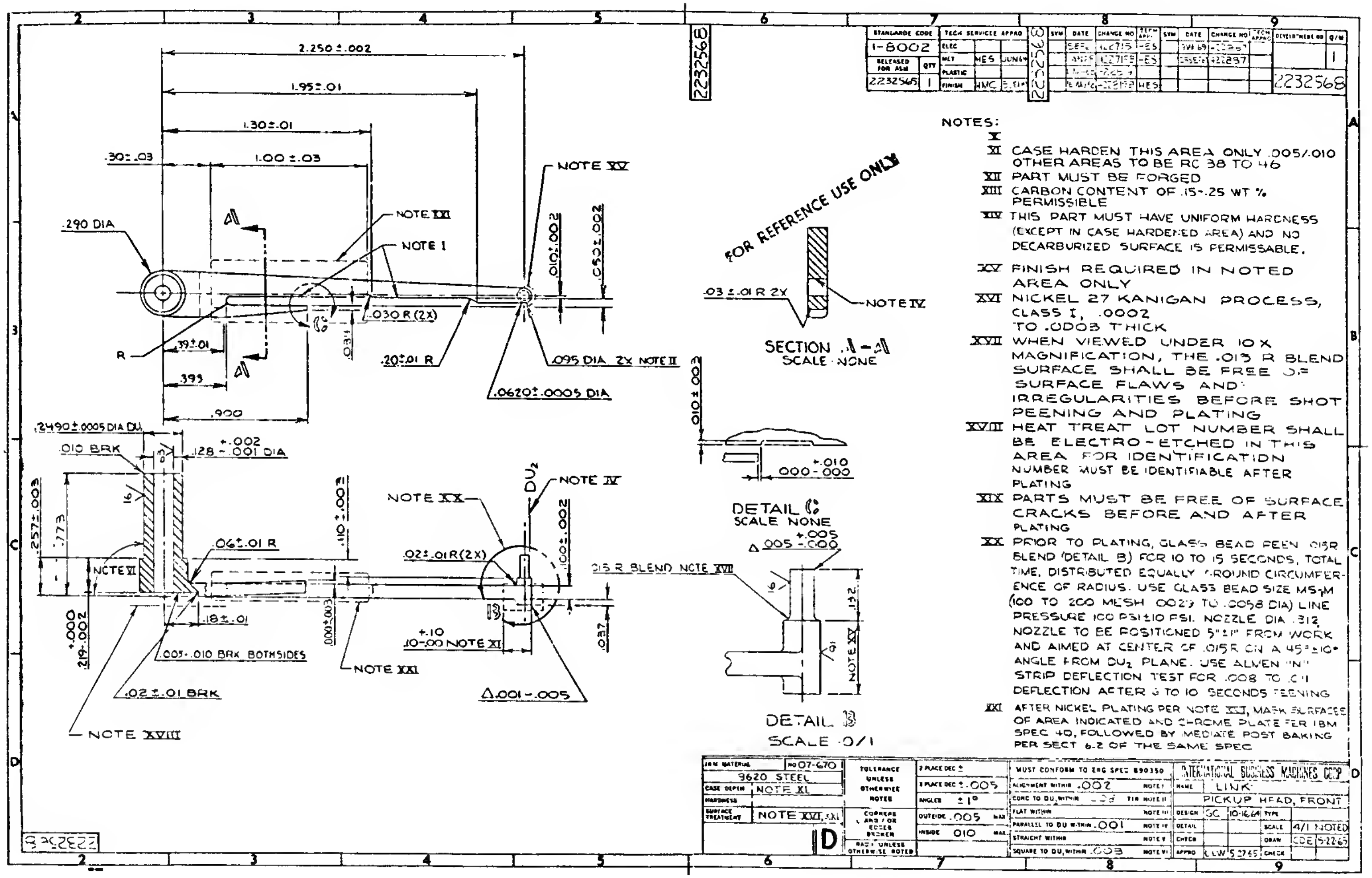


EXHIBIT B-2

Final link design.